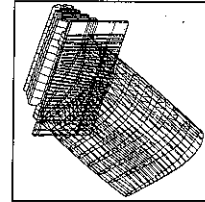


Turbomachinery Technology Seminar



Centrifugal Gas Compressor Restaging: Design Considerations



**SOLAR
TURBINES
INCORPORATED**

Contents

	Page
INTRODUCTION	1
TYPICAL OPERATION	1
DESIGN CHALLENGE	1
STANDARD PARTS PHILOSOPHY	2
INTERCHANGEABLE PARTS PHILOSOPHY	4
TYPICAL RESTAGES	4
EXPERIENCE	5
SUMMARY	6

Centrifugal Gas Compressor Restaging: Design Considerations

R.M. Wardall

Product Manager,
Saturn Gas Turbine and
Gas Boost Compressor

INTRODUCTION

The extensive use of the versatile centrifugal compressor demonstrates its suitability for almost any application in the oil and gas industry. Basically, there are two types of applications: "process", with fairly constant operating conditions as in refinery applications; and "field", with operating conditions that change considerably over time, either periodically or seasonally, as in production and transmission applications.

In field applications, it is not unusual for compressors to become candidates for restaging in order to maintain optimum performance and/or to reduce annual operating costs. This paper discusses centrifugal compressor restaging, how it is accomplished and under what conditions, and compressor design features important to restage capability.

For clarification, "restaging" is defined as the altering of a centrifugal compressor's performance envelope by changing the number of impellers or "stages" for greater or less head-making capability, changing the flow range with higher or lower flow impellers, or installing higher efficiency impellers which the original manufacturer may have developed during a product improvement program. Restaging, therefore, is not synonymous with repair, overhaul, or rebuilding, but can be accomplished during those procedures.

TYPICAL OPERATION

A given centrifugal compressor, like most types of compressors, is capable of operating over a wide range of pressure, flow, temperature, and gas conditions. There are, however, finite limits to this capability, such as those imposed by 100-percent speed and surge. There are also limits related to the desirability of maintaining optimum unit efficiency, for example, which is

of particular interest in this era of increasing fuel costs.

When operating conditions change, as in a declining pressure formation or reduced production volumes, the change can be sufficient to cause the compressor's normal operating point to move to a region of reduced efficiency, into surge, or beyond the 100-percent speed capability of the compressor.

In other cases, the light weight and portability of gas turbine compressor sets allows them to be relocated when various projects end and new ones begin. This not only extends the useful life of the equipment, but is more cost effective than simple abandonment — a typical fate of other types of compressors. However, if different operating conditions are to be encountered at the new location, the restage potential should be evaluated, since performance improvements of as much as 25 percent can be possible.

DESIGN CHALLENGE

In the early days of centrifugal compression, it was customary to design a series of frame sizes for a given line of compressors with a general impeller configuration for each size. For a given frame size, it was the general intent to custom design and fabricate the impeller or series of impellers for the specific operating point of each machine as it was sold. This approach gave rise to the practice that persists even today of focusing on the single design point. Manufacturers were reasonably certain of being able to predict and guarantee the design point, but they were less certain of the balance of the performance characteristics. Although this circumstance was acceptable for "process" applications, it sometimes created problems in production or transmission applications with their variable operating conditions. It was not

uncommon, for example, for two supposedly "identical" compressors operating in parallel, to manifest load-sharing mismatch when operating at off-design point conditions (Figure 1).

The compressor design engineer was faced with the challenge of obtaining consistently equal performance from "identical" compressors regardless of manufacturing date or changing operating conditions. The design challenge was solved by adopting a philosophy of using *standard* and *interchangeable* aerodynamic component parts. Within this philosophy, consistently equal performance could be achieved through the use of precision cast parts (Figure 2), and changing operating conditions could be accommodated by designing those parts to be interchangeable, permitting a quick and easy restage. From the manufacturer's perspective, this philosophy also supported the related objectives of quick shipment capability, low risk performance guarantees, off-the-shelf spare parts availability, and low cost.

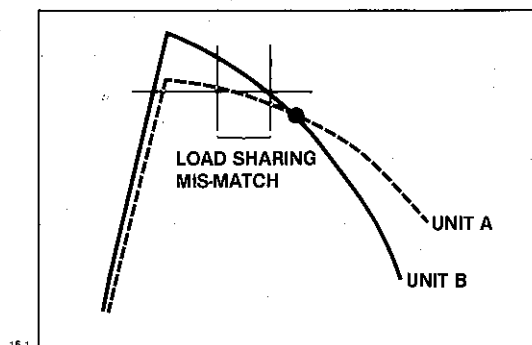
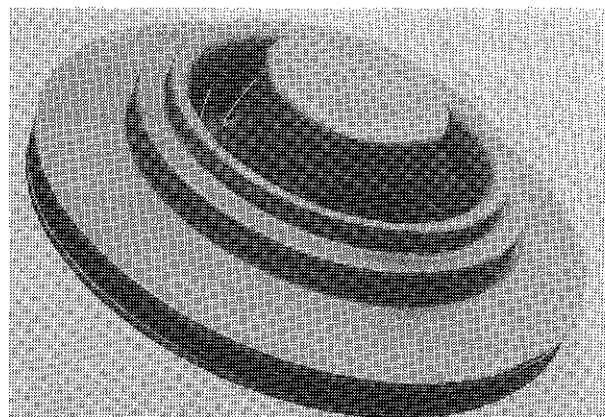


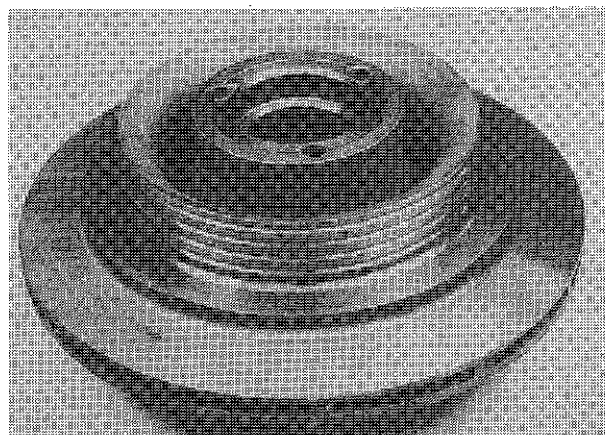
Figure 1. Two "Identical" Compressors with Common Design Point but Off-Design Loadsharing Mismatch

STANDARD PARTS PHILOSOPHY

Each compressor frame size utilizes a family of five or more impellers of different flow capacities, all having the same tip diameter but with different inlet diameters and discharge widths (Figure 3). In addition, each impeller can be equipped with one of three different turning vanes (Figure 4) which provide some amount of prewhirl to provide three different flow ranges: "with rotation" for lower flows; "against rotation" for higher flow; and "zero" for no change in flow. Solar's practice is to give each impeller in a family a letter designation "A", "B", "C", etc., and to give each turning vane a numerical



Impeller Casting Prior to Machining



Finish Machined Cast Impeller

15-2

Figure 2. Standard Aerodynamic Components

designation. Altogether, this provides a total of 15 flow characteristics across the total flow range for the given compressor frame (Figure 5). The application engineer has this array of characteristics from which to select the appropriate staging for the application requirements.

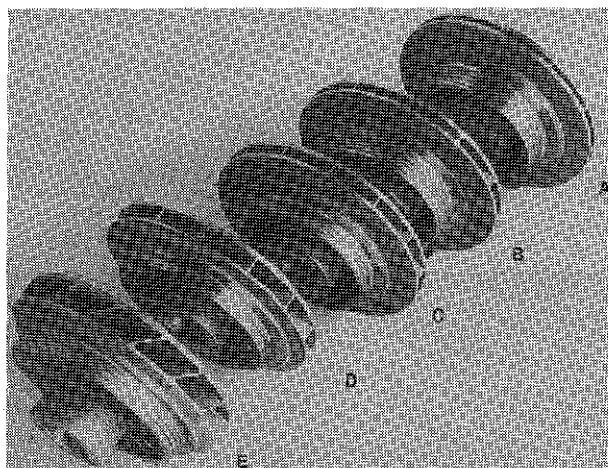


Figure 3. Family of Compressor Impellers

15-3

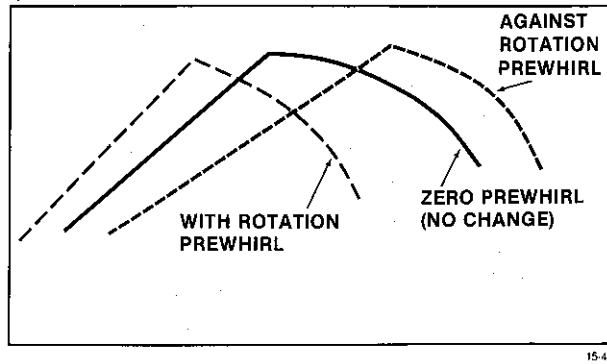


Figure 4. Effect of Turning Vanes on Impeller Performance

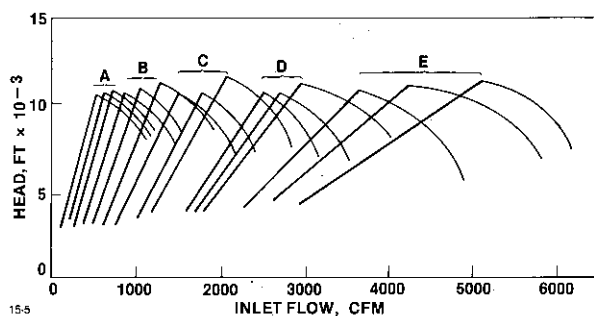


Figure 5. Compressor Single-Stage Characteristics

Using precision cast parts, which permit no appreciable variance in aerodynamic geometry, and extensive data from nearly 3000 compressors in service, the application engineer can provide highly accurate performance estimates for both new units and restages. Since identical compressors exhibit uniform performance, as shown in Figure 6, the demonstrated close correlation between calculated and actual performance using standardized aerodynamic components can result in performance estimates that easily meet normal industry tolerances.

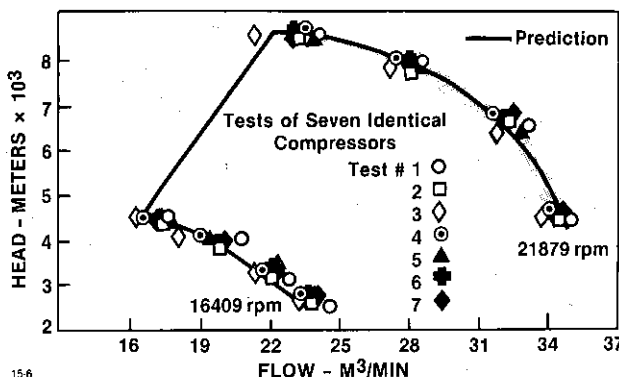


Figure 6. Compressor Test Performance

Specific Speed

Specific speed is a convenient definition to classify impellers on the basis of performance and proportion, regardless of actual size or speed. For typical centrifugal compressors with backward swept impeller blade angles, maximum efficiency generally occurs within a specific speed range of from 80 to 120, as shown in Figure 7. Establishing a minimum acceptable efficiency at 80 percent or higher of maximum efficiency will establish minimum and maximum specific speeds at approximately 45 and 180 respectively.

The definition of specific speed shows that when designing a family of precision cast impellers for a given tip diameter, a given speed, and to produce the same amount of head, an increase in specific speed will require an increase in the cross-sectional area of the gas flow passages of the impeller. Therefore, a relationship exists between the specific speed and the meridional shape of the impeller (Figure 8).

Reduction in efficiency at low specific speeds results from the relatively higher frictional losses in the narrow passages of the impeller and stationary parts, and the higher disk friction losses in the gaps between the impeller and the stationary shrouds. Efficiency reduction at high specific speed is mainly the result of high mixing losses due to a relatively

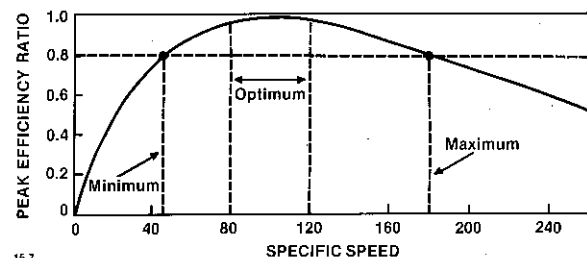


Figure 7. Peak Stage Efficiency versus Specific Speed

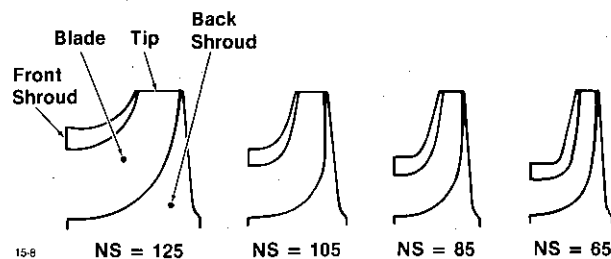


Figure 8. Relationship between Specific Speed and Shape of Impeller

small radius of curvature of the impeller front shroud and relatively high exit losses.

The designer's intimate understanding of these relationships guides the design of each impeller in the family and the companion inlet guide vanes.

INTERCHANGEABLE PARTS PHILOSOPHY

Once the aerodynamic design for the standard parts was established, the designer was faced with the task of making them interchangeable to accommodate the fast, easy restage objective.

Industry practice has traditionally used shrink-fit, solid-shaft rotor construction methods. This resulted in a somewhat permanent rotor assembly that did not readily permit the substitution of different impellers on the shaft in the event of a restage because of the high cost and difficulty in removing and reinstalling an impeller via the shrink-fit process. Because this was considered unacceptable, an alternate approach was to use a modular rotor assembly (Figure 9).

The modular rotor assembly consists of end shafts, impellers, rotor spacers to maintain a constant bearing span, and a through bolt. These components are individually balanced and are dowelled and piloted to each other for torque transmission. The entire assembly is clamped together with the through bolt. This construction provides a relatively larger effective shaft diameter which yields a stiffer, more dynamically stable rotor system in comparison to a traditional shrink-fit rotor system of the same aerodynamic capability (Figure 10).

The modular rotor assembly is also easy to disassemble. The payoff for restaging is twofold: impellers that can be reused in the

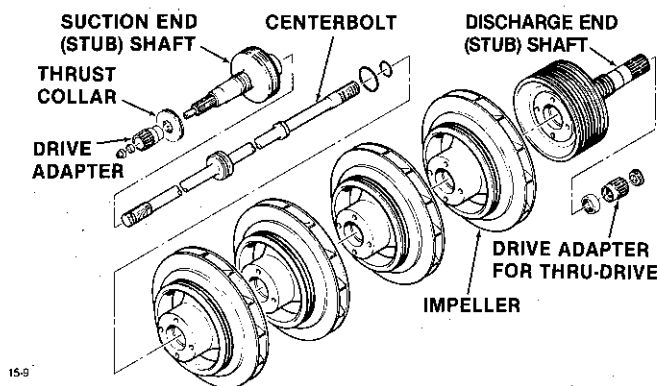


Figure 9. Typical Modular Rotor Assembly

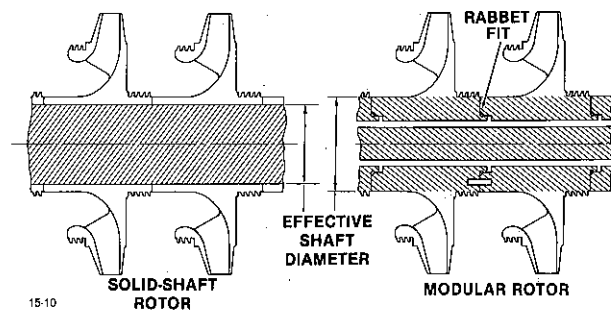


Figure 10. Comparison of Solid-Shaft Rotor with Modular Rotor

restaged rotor are easily salvageable; and downtime is minimized. Reusing the impellers instead of purchasing new impellers for the new performance condition, enhances the economic feasibility of restaging to maintain optimum compressor performance and the lowest possible operating costs.

TYPICAL RESTAGES

A typical restage candidate is a compressor in a gas gathering application where the discharge pressure is to remain constant at 800 psia despite a suction pressure that is expected to decline below 500 psia. Because a higher compression ratio is required, more impellers are needed. Based upon the prospective future operating conditions, which may see well-head pressure decline to as low as 325 psia in a few years, the application engineer identifies that the current two-stage rotor needs to be restaged to a four-stage rotor. Depending upon the pressure/volume relationship at the lower suction pressure, lower or higher flow impellers may be needed. Relative performance curves are shown in Figure 11.

The existing two-stage rotor (Figure 12) has one "D" impeller and one "C" impeller. The restaged rotor will have four "C" impellers so the existing "C" impeller, subject to inspection, can be used in the new four-stage rotor assembly. The "D" impeller becomes surplus and three new "C" impellers are required.

The interchangeable parts philosophy and the modular rotor system allow the old impeller to be used anywhere on the shaft, since all the impellers are mechanically interchangeable. The "new" impellers can be sourced from a user's spares inventory, possibly left over from previous restages, or from the manufacturer's new or reconditioned parts inventories. Any surplus impeller can be returned to the user's

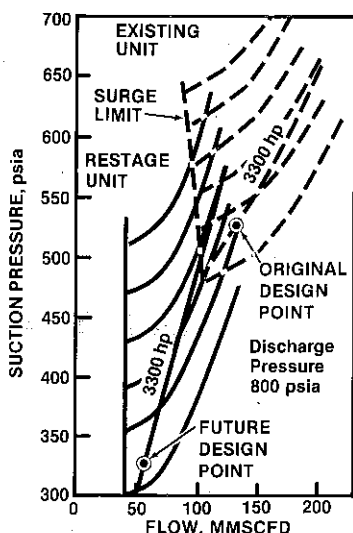


Figure 11. Restage Proposed for Lower Suction Pressure

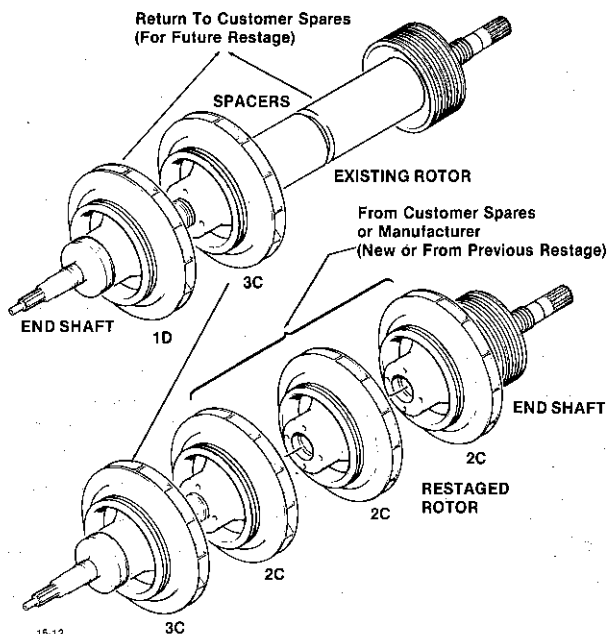


Figure 12. Versatility of Modular Rotor Using Standard Interchangeable Aerodynamic Component Parts

spares inventory or for credit where that option is available.

Another example is a case shown in Figure 13 where production has declined significantly over a period of several years with flow reducing from point A to point B. The existing compressor, however, is actually operating at flow point C by using a bypass to recycle a portion of the flow back to the compressor inlet in order to maintain minimum flow and avoid compressor surge. Recycling a centrifugal compressor is a

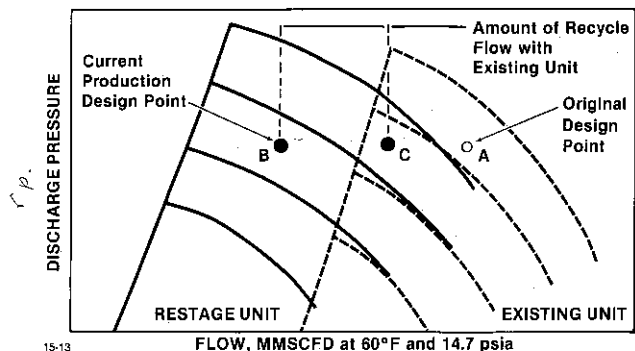


Figure 13. Restage Proposed for Lower Flow Condition

common practice, but can be costly since it represents wasted horsepower and fuel. It is economically acceptable for short-term, off-design conditions, but not for continuous operation.

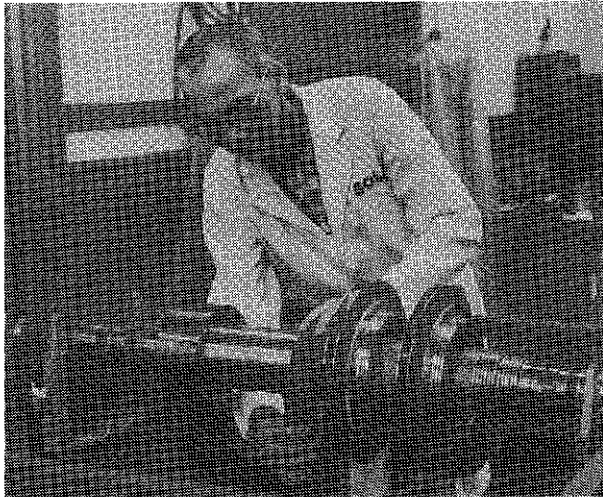
In this case, a restage to lower flow impellers would eliminate the recycling and, depending upon exact horsepower, flow, and fuel costs, the restage cost could be paid back within several months. In fact, it is not uncommon for users to obtain annual fuel cost savings of 10 to 30 percent after this type of restaging.

Where several compressors are involved in a restage program due to a field pressure decline, careful planning can permit maximum reuse of the existing interchangeable components by allowing them to be relocated quickly from one compressor to another to limit downtime.

Depending upon the user's need and preference, the restages in these examples could be performed at any of several places, including the manufacturer's overhaul or repair center, the user's shop, or at the installation site. Field restage requires shop assembly and balancing of the new rotor (Figure 14). The rotor is then match-marked for reassembly at the time of the installation site restage. In some cases, special restage tools may be required. They will have been engineered, proofed, and made available in special field tool sets for the particular compressor models involved in the restage.

EXPERIENCE

Today, several manufacturers including Solar employ the modular gas compressor rotor design because of the advantages previously mentioned. Solar has extensive experience in restaging many of the fleet of nearly 3000 compressors shipped since 1960. Table 1 documents the amount of restage activity through January 1, 1984.



15-14

Figure 14. Multi-Plane Balancing

Compressors which operate with "summer" and "winter" rotors are included in the "one restage" category, but are actually restaged annually. Thus, the absolute number of restages is larger than the total shown.

SUMMARY

The design philosophy of standard and interchangeable aerodynamic component parts inherent in the modular rotor system yields con-

Table 1. Solar's Restage Experience through 1/84

Units Restaged	720	(24.5% of total shipped)
No. of Restages	954	(32.4% of total shipped)
Restages/Unit	Units	Total Restages
One Restage	533	533
Two Restages	147	294
Three Restages	34	102
Four Restages	5	20
Five Restages	1	5
Totals	720	954

15-15

sistent, predictable performance and facilitates restage capability. The experience of nearly 1,000 restages highlights the achievement of two major objectives: maximized equipment performance and utilization, and minimized equipment operating and support costs. While these objectives were established long ago, their underlying spirit remains today and will continue into the future as the value of turbomachinery continues to increase for the user.